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MOTION CONTROL OF ELECTRO-PNEUMATIC SYSTEM BASED ON DIRECTIONAL CONTROL SOLENOID VALVE

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ABSTRACT

This work is concerned with the design and implementation of position control electro-pneumatic system using ON/OFF solenoid directional control valves. To realize this approach, it's required to set up a laboratory mechanical test rig involving pneumatic, mechanical and electrical apparatus. The proposed design of control system includes electro-pneumatic circuit interfacing with position sensor, especially used in this work to detect the distance of the rod of pneumatic actuator. The suggested model mainly consists of double acting pneumatic cylinder, two solenoid directional valves, two position sensors and electrical control circuit. The simulation process of the employ system has been done with using MATLAB software and the obtained theoretical as well as practical results have been compared to investigate the pest performance. The obtained results showed that the proposed system can be applied with high accuracy and efficiency.

KEYWORDS: Electro-Pneumatic System, MATLAB & Control Valves

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Original Article

1. INTRODUCTION

In the last decades, the pneumatic systems are extensively applied in different filed of industrial application due to rapid increasing in automation process**[1]**. They have many advantages over than other control system. Some of these can be provide important advantages are easy and ready availability of air, very simple construction, clean and low cost **[2, 3]**. Also, pneumatic systems are especially can be easily maintenance and provide high speed control action with high power to weight ratio **[4]**. It can be noted from characteristic of pneumatic system that the response of speed control stability system can be achieved when using the appropriate electro- pneumatic component**s [5, 6]**. Many researchers are dealt with control application of the electro-pneumatic system. Such as studded of magneto rheological technology of pneumatic motion control system the tolerance of the stop point position was very small at fastest velocities**[7].** The design of position control system for pneumatic actuators based on the sliding-mode control technique show the steady state error was very small and the response of the pneumatic actuator according to the desired position control system was very accurate **[8]**. The studied of force control application of a pneumatic actuator according to the fact that the delivered power from a pneumatic actuator s showed that the ability to convert a pneumatic actuator into a force generator**[9]**. The studied of pneumatic motion control system using on/off solenoid valve with a position feedback sensor show that the performance was very well and low steady state position errors. **[10]**. The suggested method of position and force control on pneumatic cylinder of robots with nested control loop can be resolve the problem of the supporting legs

of legged robots **[11]**. The investigation of servo pneumatic position control system with using two control algorithms showed the sliding mode control algorithm has better performance in comparison with algorithm that using position plus velocity plus acceleration feedback that combined with feed forward and dead zone compensation **[12]**. The studied of position control of pneumatic actuator using novel pulse width modulation system with pulsing on/off directional control solenoid valves added of proportional integral derivative (PID) controller provide significant improvement in the results of the system response **[13]**.

In this work, the attempt of design and implementation of position control electro-pneumatic system using ON/OFF solenoid directional valves has been carried out. The desired approach includes set up laboratory mechanical test rig. The prototype electro-pneumaticis proposed control system interfacing with position sensor to detect the stroke position of pneumatic cylinder. The simulation process of the suggested system has been done using MATLAB program software and the obtained theoretical results have been comparing with practical results. The organization of this paper has been arranged as follows; the second section has included the description of the electro-pneumatic motion control system while the modeling of electro-pneumatic position control system is presented in the third section, the experimental and theoretical results has been introduced in the fourth section, followed by conclusions.

2. THE ELECTRO-PNEUMATIC MOTION CONTROL SYSTEM

The proposed electro-pneumatic position control system mainly consists of double acting pneumatic cylinder and two solenoid directional valves. The pneumatic system will be supplied by pressurized air provided from air compressor unit. Some accessories such as conditioning unit (FRL), manual variable throttle valves, pressure gages, silencers and air flow meter are used. The position control of pneumatic required two position sensors and electrical control circuit as shown in figure 1.

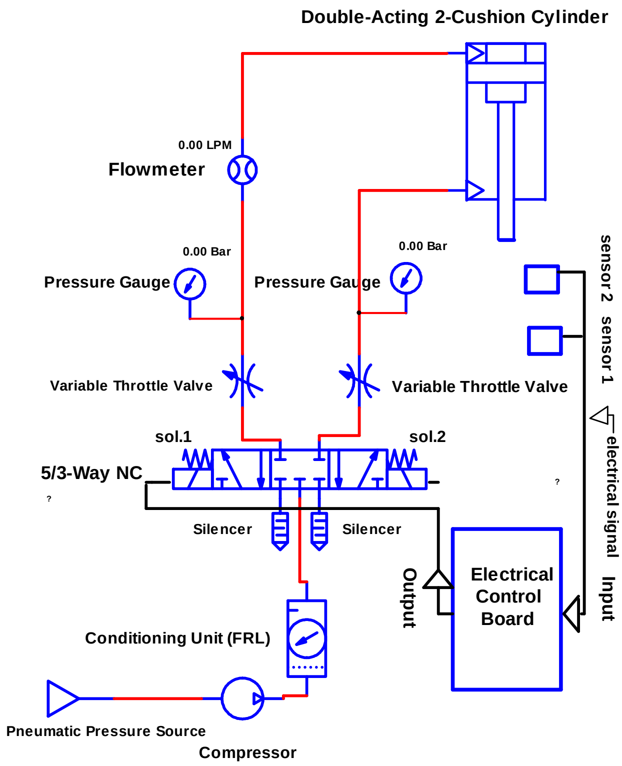

Figure 1: The Layout of Electro-Pneumatic System [Introduced by Researchers]

The electro-pneumatic system can be subdivided into power and control subsystem. The power system contains air compressor driven by electrical motor to supply the required pressurized air to pneumatic cylinder throw pipes and hoses. The control unit include 5/3 way directional control solenoid valve and electrical control board interfacing with two electro-mechanical position sensors. The first sensor has been placed at a distance (228 mm) measured with respect to initial position while the second sensor placed at a distance (37 mm) as shown in figure 2.

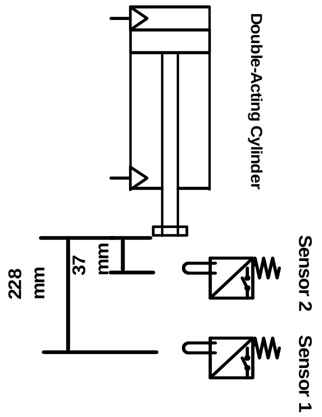

Figure 2: The Placed of Electro-Mechanical Position Sensors

The electrical position control circuit contains three main switch types (ON/OFF), two electro-mechanical position sensors, two solenoid coils, two solenoids signal and two relay. Three practical steps to obtain the required control action either in downward or upward motion of piston rod of cylinder are shown in figure 3.

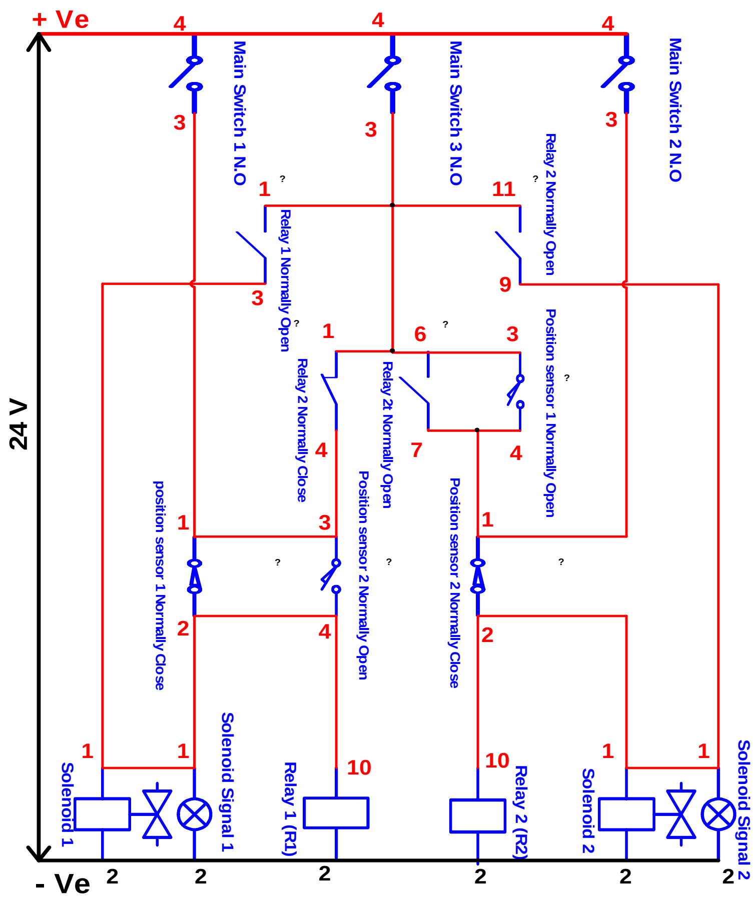

Figure 3: The Layout of Electrical Position Control Circuit

The activated and inactivated solenoid coils was depending on the electro-mechanical state of sensors. If the main switch 1 turned on, the solenoid coil 1 would activated and the piston rod of cylinder downward moving from initial point to distance equal to set point 1 (228 mm). As piston rod of cylinder reach sensor 1, the electrical signal will cut off and solenoid 1 inactivated causing stop motion of piston rod cylinder. If main switch 2 turned on, the solenoid 2 will be activated and the piston rod cylinder moving from set point 1. The piston rod cylinder still in motion at direction upward until reaching to set point 2witches at (37 mm). The position sensor in set point 2 will be responsible to stop the motion of rod. If turn on the main switch 3 in order to get repeated motion (upward and downward) the piston rod cylinder will be limited in stroke set point 1 at (228 mm) and set point 2 (37 mm).

3. MODELING OF ELECTRO-PNEUMATIC POSITION CONTROL SYSTEM

The mathematical model of the electro-pneumatic position control system has been derived containing the double action cylinder and the directional solenoid control valve to get the transfer functions of there. The modeling of pneumatic cylinder represented by the figure 4.

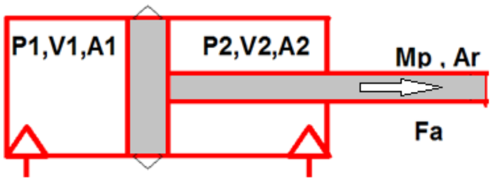

Figure 4: The Schematic diagram of the Double Acting Cylinder

The required equations of motion for pneumatic cylinder can be calculated with applying the Newton second low:**[14, 15]**

$$\sum F = M.a \tag{1}$$

For downward motion

$$M\ddot{Y} + \beta\dot{Y} + F_f + Mg = P_1A_1 - P_2A_2 - P_aA_r \tag{2}$$

For upward motion

$$\ddot{Y} + \beta\dot{Y} + F_f - Mg = P_2A_2 - P_1A_1 - P_aA_r \tag{3}$$

Where:

M: Piston rod mass

β: Viscous of friction

F_f: Friction force

G: gravity

P_1: Pressure cylinder in chamber 1

A_1: Piston area in chamber 1(downward stroke)

P_2: Pressure cylinder in chamber 2

A_2: Piston area in chamber 2 (upward stroke)

P_a: Atmosphere pressure

A_r: Rod area

The mathematical transfer function of the cylinder at full stroke can be obtained with assumed air inside the cylinder is ideal gas, the pressure of air sources is constant and no leakage in the cylinder. Then, the mathematical model of pneumatic cylinder will be as follows **[16, 17]**

$$\frac{y(s)}{x(s)} = \frac{C_x/A_p}{s(\frac{1}{w_n^2}s^2+\frac{2\zeta s}{w_n}+1)} \tag{4}$$

Where:

Y_{out}: The output position of the pneumatic cylinder.

X_{sp}: The spool displacement of control valve.

ζ : damping ratio

w_n : Natural frequency

The natural frequency and damping ratio for full stroke will be:

$$w_n = \sqrt{\frac{2*A^2*K_a*p_i}{M*V_c}} \tag{5}$$

$$\zeta = \frac{c_f}{2*A}\sqrt{\frac{V_c}{2M*K*P_i}} \tag{6}$$

Where:

K: the ratio of specific heat of air

M: total mass

V_c : Volume of cylinder

c_f : Viscous friction cylinder

P_i : Pressure inside cylinder (practically measurement)

The solenoid directional valve can be modeled to three subsystems electrical, mechanical and electromagnetic.

The equation of motion of mechanical and electromagnetic subsystem can be written as follows:**[14, 15]**

$$m\ddot{x} = F_{mag} - c\dot{x} - Kx \tag{7}$$

$$m\ddot{x} + c\dot{x} + Kx + F_f = K_c i \tag{8}$$

Taking Laplace transformation with assumption the initial conditions are zero and neglected coulomb friction force (F_f) presented as:

$$ms^2x(s) + csx(s) + Kx(s) = K_c i(s) \quad (9)$$

Where:

x= position of spool

K=constant of spring

c= coefficient of damping

K_c = coefficient of coil force

i= current of coil

The mathematical model for directional control valve will be as follows:**[17]**

$$\frac{X_{sp(s)}}{I_{(s)}} = \frac{K_c}{ms^2+cs+K} \quad (10)$$

The mathematical model of the solenoid coil can be obtained with Appling Kirchhoff's voltage law (KVL):

$$Vs = Vr + Vi \quad (11)$$

The transfer function of electrical part (solenoid coil) written as:

$$Ls.i(s) + R.i(s) = V(s) \quad (12)$$

Where:

L = inductance of the magnetic coil

V= applied voltage

R = the resistance of electromagnet coil

i = the current of electromagnet coil.

The transfer function (12) can be written as:**[17]**

$$\frac{I_{(S)}}{V_{(s)}} = \frac{1}{Ls+R} \quad (13)$$

The air flow rate passing through the directional valve due to displacement of the spool writing as:**[16, 17]**

$$Q = Cx * Xsp \quad (14)$$

4. THE EXPERIMENTAL AND SIMULATION RESULTS

The experimental work had been done in control laboratory of mechanical department in collage of engineering of Al-Mustansiriyah University. It was continued building mechanical test rig and implementation electro-pneumatic system driving by electrical motion control unit. In the setup, the mechanical test rig depends on the characteristic of pneumatic cylinder (piston diameter, stroke). The double acting pneumatic cylinder and directional solenoid valve have been characteristic as shown in table 1.

Table 1: The Characteristic of Cylinder and Solenoid Directional Valve

Name	Characteristic	
Double Acting Cylinder	Piston area	Ap=8.04*10^-4 m^2
	Viscous coefficient	C=100N(m/s)
	Piston rod mass	mp=0.532 Kg
Directional Control Valve	Voltage	V=24VDC
	Resistance	R=230.77 Ω
	Inductance	L = 0.98 H
	Damping Coefficient	Csp=0.01 N/(m/s)
	Spring Stiffens	k=75 N/m
	Mass Spool	mp=10 g
	Mechanical Converter	Kco =1.3 N/A
	Spool displacement	Xsp=2 mm

The pneumatic cylinder placed vertically to provide position control system. The electrical control unit included all electrical components as shown in figure 5.

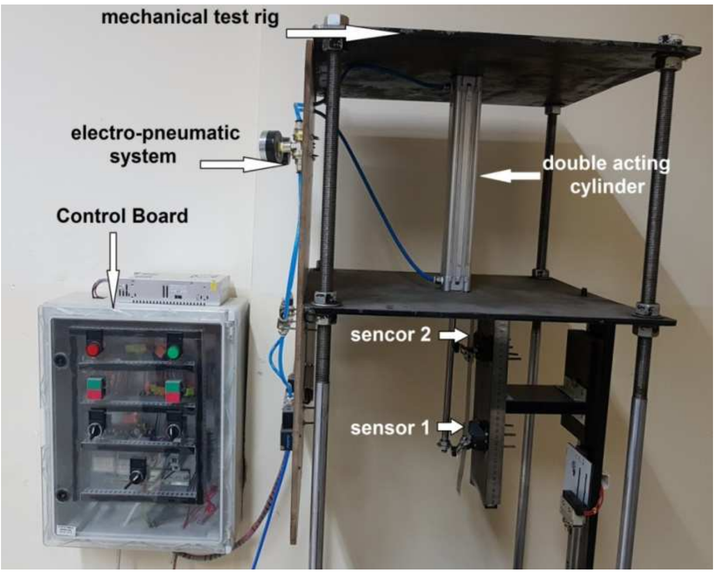

Figure 5: The Employed of Mechanical Test Rig with Electro-Pneumatic System and Control Board

The desired position sensors have been placed horizontally in forward path of the piston rod cylinder. Mechanic cam has been connected in the tip of piston rod cylinder to ensure better contact, occurs with the roller of the sensors in downward and upward motion as shown as in figure 6.

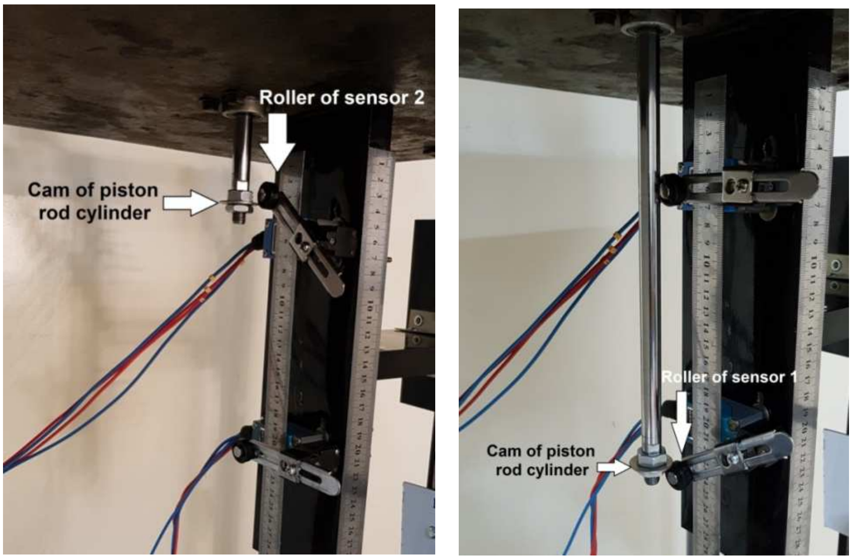

Figure 6: The Contact between Piston Rod Cylinder and Mechanical Part of Position Sensors 1&2

The experimental results of piston rod cylinder position in downward and upward motion listed in table 2.

Table 2: The Calibration of Position Control in Downward and Upward Motion of Pneumatic Actuator

Air Flow Rate (Q) L/M	Pressure (P) Bar	Piston Rod Actuator Position		Tolerance mm
		Stop Point at Set Point1 (228mm)	Stop Point at Set Point2 (37mm)	
3	1.00	228	37	0
5	1.20	228	37	0
7	1.25	228	37	0
10	1.45	229	36	1
13	1.60	229	36	1

The experimental stop position time of electro-pneumatic position control system in downward motion with full stroke (250 mm) is as listed in table 3.

Table 3: The Experimental Stop Position Time, Velocity and Pressure of Pneumatic Actuator Results

Flow Rate (Q) L/M	Stop Position Time (t) sec	Experimental Velocity (V) m/s	Experimental Pressure (P) Bar
3	8.3	0.03	1
5	5	0.05	1.2
10	2.43	0.102	1.45
13	1.8	0.139	1.6

The relationship between the experimental stop position time and air flow rate are shown in the figure 7.

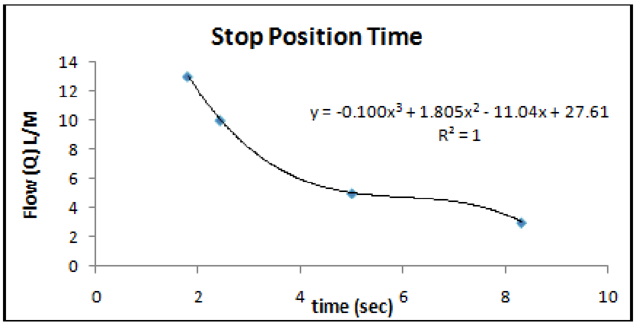

Figure 7: The Relationship of Air Flow Rate with Practical Settling Time

The simulation model of electro-pneumatic position control system has been done by MATLAP program software with using proportional controller and unity feedback for full stroke as shown in figure 8.

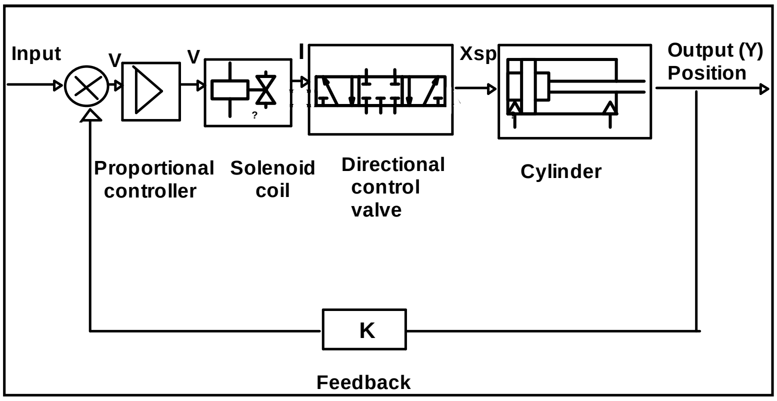

Figure 8: The Simulation Model of Electro-Pneumatic Position Control System

To simulate the electro-pneumatic position control system with the characteristic of pneumatic cylinder and solenoid directional control, according to equations 4, 5, 6, 10, 13 and 14, then

$$\frac{I_{(S)}}{V_{(s)}} = \frac{1}{0.98s+230.77} \qquad \text{Solenoid coil}$$

$$\frac{X_{sp(s)}}{I_{(s)}} = \frac{1.3}{0.01s^2+0.01s+75} \qquad \text{Directional control valve}$$

$$\frac{y(s)}{x(s)} = \frac{C_x/_{8.04*10^{\wedge}-4}}{s(\frac{1}{w_n^2}s^2+\frac{2\zeta s}{w_n}+1)} \qquad \text{Cylinder}$$

$$C_x = \frac{Q}{0.002} w_n = \sqrt{0.017 * Pi}\zeta = 62189.05\sqrt{\frac{1.35*10^{\wedge}-4}{P_i}}$$

The step response of cylinder rod was displayed from running the model of electro-pneumatic position control system with using proportional controller (Kp) equal to (230), and different air flow rate as shown in figure 9, and the simulation results can be listed as shown in table 4.

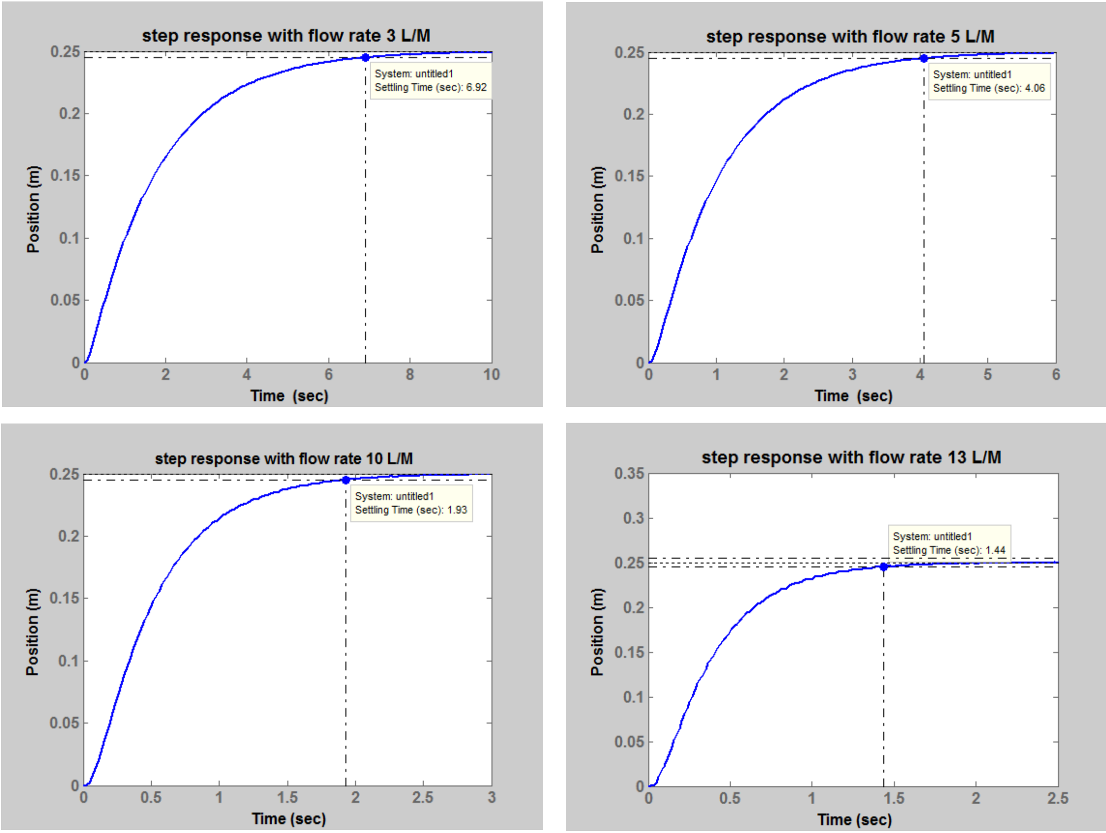

Figure 9: The Step Response of Electro-Pneumatic System at Different Flow Rate Steps

Table 4: The Simulation Settling Time Results

Step	Flow (Q) L/M	Simulation Settling Time (t) sec	Pressure (P) Bar
1	3	6.92	1
2	5	4.06	1.2
3	10	1.93	1.45
4	13	1.44	1.6

5. CONCLUSIONS

The objective of this work is to provide electro-pneumatic system motion control based on inexpensive ON/OFF solenoid directional valve with tow electro-mechanical position sensors. It's done based on the following remarks:

- Two electrical control circuit steps had been done on electro-pneumatic system to obtain the desired motion control. The proposed of first step was to stop piston rod cylinder at desired set point (at position sensor) and second to get repeated motion control.
- The experimental results showed, the piston rod cylinder stop at desired set point without tolerance when the air flow rate value in range (0-7 L/M) while tolerance accrue equal to (1 mm) when the air flow value in rang (7-13 L/M).
- The simulation results with proportional controller and unity feedback, showing the step response becoming faster when the air flow rate increase in range (10-13 L/M).

- The comparison between the experimental stop position time and theoretical settling time results for full stroke of pneumatic cylinder, showing very small tolerance value between the practical and theoretical, and the tolerance can be decreased when the air flow rate and the pressure inside cylinder increases.
- The obtained results show the high accurate performance of the proposed model.
- The effect of temperature has been neglected because, the atmospheric temperature was normal at work.
- It can be concluded that the air filter was very important to get smooth and fast performance of system with desired results.

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